

Automation of Analysis And Structural-Parametric Synthesis of Planetary Gearboxes With Two Degrees Of Freedom

D. Volontsevich^{1, a}, S. Pasechnyi²

1 – Doctor of technical sciences, professor, Transport Machine Building Faculty, National Technical University "Kharkov Polytechnic Institute", Kharkov, Ukraine

2 – Graduate student, Transport Machine Building Faculty, National Technical University "Kharkov Polytechnic Institute", Kharkov, Ukraine

a – vdo_khpi@ukr.net



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ABSTRACT. The planetary gears in transmissions of non-rail ground transport vehicles are widely used. Mostly, all modern automatic transmissions are based on the planetary mechanisms. The design of the optimal schematic diagram and the calculations of the parameters is extremely complicated. Moreover, this process is automated particularly. The design of kinematic schemes, which based on the selected linkages is the most complicated process for automatization. The algorithm and software for analysis and structural-parametric synthesis of planetary gearboxes with two degrees of freedoms is considered. The suggested approach can be applied to elementary planetary gears and systems with two or more satellites and double-wheel gears. The software is implemented only for planetary gearboxes with two degrees of freedom. The software for the planetary gearboxes with three degrees of freedom is implemented at the nearest time. The quantitative criteria of quality for estimation of planetary gearboxes quality are developed. These criteria allow to numerically evaluating the following characteristics of planetary gearboxes as the presence and magnitude of the circulating power, the relative magnitude of the torque on brakes and clutches, the relative speed of rotation of the central units, satellites and open-loop brakes and clutches and much more. The generated kinematics schemes are compared according to selected multiple criteria or integrated criteria with weights coefficients.

Introduction. Planetary transmissions are used in the power transmission of self-propelled vehicles from the first decades of the twentieth century.

At the beginning the use of planetary gearboxes has been extremely rare. This is due to the absence at that time a well-developed theory of planetary gears and a relatively low technological level of mechanical engineering.

Later the use of planetary gearboxes is continuously increased, and by the turn of XX-XXI centuries, has become almost an absolute status.

Now planetary gears are used as part of the power transmission of the majority of samples of cars and trucks, buses, industrial and agricultural tractors, construction and road vehicles, armored vehicles and other types of self-propelled vehicles of the land for various purposes.

Planetary gears an extremely widely used not only planetary gears in transmissions, but also in other transmission units – the mechanisms of rotation of tracked vehicles, in gears for the separate or the summation of power, in transmissions power take, in a distributing boxes, in the board and wheel gear units, etc.

There is every reason to believe that in the foreseeable future, the level of use of planetary gearboxes, subject to their further improvement will only increase [1].

After the traction calculation and determining of gear ratios in the gearbox there is a problem before the designer of choice the most rational scheme of its design, that implements prescribed ratios. The choice of the kinematic scheme of planetary gearboxes is the most difficult and important task for the

designer as for given ratios can build a large variety of different schemes. These schemes will differ in complexity, efficiency size, and a number of other factors affecting the level of technical design [2].

This work is dedicated to the automation of the design process of planetary gearboxes for substantial savings of time and labor.

Analysis of recent achievements and publications. Now the designers use a method of synthesis of the planetary gearboxes, which was developed by M.Kreines. His followers [1-9] have repeatedly modified this method. For a planetary transmission with two degrees of freedom, this method is well developed and is easily automated, but it contains one point, requiring direct human intervention. This stage of testing the possibility of building a kinematic scheme of a given block diagram. The problem is that not every block diagram can be implemented as a kinematic scheme because not possible to carry out communication between the elements of planetary mechanisms and controls.

Of course, similar problems leading developers of planetary gearboxes have been resolved, but the available literature there is not comprehensive information on this issue, especially since there is no available software products that allow us to solve these problems.

The aim and problem statement. The purpose of this paper is to construct an algorithm and software for the automated synthesis of planetary gearboxes with two degrees of freedom on the set gear ratio.

To achieve this goal it is necessary:

- to carry out the generation of all possible combinations of connections between elements of the planetary transmission, that could constitute scheme;
- to determine the number and position of the controls - clutches and brakes;
- to calculate the internal gear ratio of each planetary mechanism, as well as basic kinematic and power characteristics of planetary gearboxes;
- to carry out culling of schemes at the limitations associated with the value and sign of the internal gear ratios;
- to check out the possibility of constructing kinematic schemes remaining after culling;
- to calculate numerical quality criteria;
- to sort resulting schemes in selected quality criteria.

The algorithm generating all possible combinations of connections between elements of the planetary gearboxes to create the block diagram of transmission. The initial data for the solution of the problem are the values of gear ratios for the proposed planetary transmission, which are not equal to 1. In the realized software product of their number cannot exceed four. Later, when the software product can operate with a planetary transmission with three degrees of freedom, as input data must be entered the degrees of freedom of the transmission.

We used the original algorithm for constructing all possible combinations of connections for planetary gearboxes. Those schemes that do not meet the criteria for synthesis rejected.

An algorithm for constructing all possible links combinations:

1. Determination using table 1 minimum number of planetary gears and the necessary controls.
2. Determination of the minimum number of permanent links between the elements of planetary gears at a given number of degrees of freedom is performed by using the formula:

$$N = 2n_{EPM} - W, \quad (1)$$

where N – number of permanent links between the elements of planetary gears;

n_{EPM} – the number of elementary planetary mechanisms;

W – number of degrees of freedom.

3. Enumerating all the possible options for connections between the elements of planetary gears:

- input link in turn is connected to all elements
- output link in turn is connected to all the remaining elements;
- for the selected input and output checks all combinations of permanent links.

4. For each of the resulting combinations of links determined by elements, which can be connected to the brakes;

5. Rejection of options is carried out according to the following criteria:

- not allowed a direct connection between input and output;
- braking is not permitted input and output shafts;
- not allowed constant communication between elements of the same planetary mechanism;
- not allowed connection of element which is connected to the drive shaft, with elements belonging to the previous planetary rows;
- not allowed connection of element which is connected to the driven shaft, with elements belonging to the previous planetary rows.

Table 1. The minimum number of structural elements of the planetary gearboxes [1]

number of degrees of freedom W	The main elements of the planetary gearboxes	Number of elements in the planetary gearboxes with the number of gears forward and reverse (without direct transfer)										
		1	2	3	4	5	6	7	8	9	10	11
2	locking clutch – n_{LC}	1	1	1	1	1	1	1	1	1	1	1
	stopping brake – n_{SB}	1	2	3	4	5	6	7	8	9	10	11
	elementary planetary mechanisms – n_{EPM}	1	2	3	4	5	6	7	8	9	10	11
3	locking clutch – n_{LC}	–	2	2	2	2	2	2	2	2	2	2
	stopping brake – n_{SB}	–	2	1	2	2	3	3	3	3	4	4
	elementary planetary mechanisms – n_{EPM}	–	1	2	3	3	4	4	4	4	5	5

The algorithm for determining the internal gear ratios of the planetary sets according predetermined gear ratios of planetary gearboxes and a full analysis of the scheme.

In any planetary transmission the speed of all central elements are interconnected via the internal gear ratios. For each planetary mechanism can be written Willis's equation:

$$\omega_a - k\omega_b + \omega_h(k-1) = 0, \quad (2)$$

where k – internal gear ratio of planetary mechanism, which for simple planetary mechanisms is defined as the ratio of the number of teeth on the crown gear and the sun gear.

Determination of internal gear ratios of the planetary mechanisms according to known gear ratios of planetary transmission is associated with the solution of the system of variable structure. It was therefore decided to find internal ratios for all planetary mechanisms by searching in the specified range with the selected step and provide the necessary precision. If the solution in a specified range with satisfying accuracy, has not been found, then this scheme is eliminated because of the impossibility to implement the specified gear ratios.

Further, in accordance with the found internal gear ratio determines the lowest possible number of teeth on all gears under the terms of alignment, assembly and neighborhood. After that, we spend a full kinematic and power analysis of scheme according to known methods of synthesis of planetary gears [1-6].

It includes:

- Determining the relative rotational speed of central links of the planetary transmission and satellites in all gears, provided that the rotational speed of the drive shaft $\omega_{IN} = 1$;
- Determination of the relative torques on all central links of the planetary transmission and elements of controls without losses in engagement with the proviso that $M_{IN} = 1$;
- Determination of the relative torques on all central links of the planetary transmission and elements of controls, taking into account losses in engagement with the proviso that $M_{IN} = 1$;
- determine quantitative of quality criteria [11];
- determining the integral quality assessment of the scheme [11].

The algorithm for determining the quantitative criteria of quality planetary gearboxes.

To automate parametric synthesis of planetary gearboxes is not enough to generate all possible schemes, also need to quantify qualitative criteria, and only choose the best scheme. Defining quality criteria is performed based on data obtained from kinematic and force calculations.

The proposed system of criteria for quality schemes of planetary gearboxes is not meant to completely objective and comprehensive assessment of the actual transmission as a technical product. This system allows you to perform a comparative assessment of different planetary gearboxes on the schematic level, assess the degree of perfection of the kinematic scheme. These simple and rather obvious criteria, however, possible to obtain a quantitative estimate of the most important indicators of the analyzed kinematic scheme in a form suitable for comparison of the several variants of planetary gearboxes under selecting the best out of the scheme under consideration.

To determine the quantitative criteria of quality of planetary gearboxes were taken as the basis of criteria given in [1]:

1) The criterion of the power transmitted by the elements of the planetary transmission on the forward gears.

This criterion estimates the amount of power transmitted by the element of the planetary transmission as a fraction of the input power. The value of this criterion is determined by the expression:

$$K_1 = \frac{N_{FG \max}}{N_{IN}}, \quad (3)$$

where $N_{FG\max}$ – maximum power transmitted most stressful element the planetary gear to the forward gears;

N_{IN} – the power supplied to the planetary transmission.

2) The criterion of the power transmitted by the elements of the planetary transmission on the reverse gears.

This criterion has the same meaning and is determined as well as the previous one, only for reverse gears:

$$K_2 = \frac{N_{RG\max}}{N_{IN}}. \quad (4)$$

3) The criterion of the loading of the locking clutch.

This criterion reflects the value of the maximum torque transmitted by included locking clutch, and allows roughly estimating the axial and radial dimensions of the clutch. The value of this criterion is determined by the expression:

$$K_3 = \frac{M_{LC\max}}{M_{IN}}, \quad (5)$$

where $M_{LC\max}$ – maximum torque transmitted included locking clutch;

M_{IN} – torque supplied to input of the planetary transmission.

According to the analysis of many real schemes of planetary gearboxes can be considered optimal criterion values in the range 0.6 ... 0.8.

4) The criterion of the loading of the stopping brake.

This criterion reflects the value of the maximum torque transmitted by included stopping brake. The value of this criterion is determined by the expression:

$$K_4 = \frac{M_{SB\max}}{|i_{PGB}|}, \quad (6)$$

where $M_{SB\max}$ – maximum torque transmitted included stopping brake;

$|i_{PGB}|$ – the absolute value of the transmission ratio of the planetary transmission on the transfer for which the specified torque $M_{SB\max}$.

It should be noted, that this criterion makes it impossible to estimate the size of the required brake that are included at different levels of transmission.

5) The criterion that determines the speed of rotation of the bearings placed in satellites.

This criterion reflects the value of the maximum speed of the "high-speed" satellite of the planetary transmission. It determines the reliability and durability of bearings satellites. The value of this criterion is determined by the expression:

$$K_5 = \frac{|\omega_{satmax}|}{|\omega_{max}|}, \quad (7)$$

where ω_{satmax} – maximum relative angular velocity of the satellite in the planetary set which is loaded with non-zero torque;

ω_{max} – the angular velocity of the high-speed main link in the planetary mechanism, which was defined the maximum relative angular velocity of the satellite and on the same transmission.

Criterion K_5 values for various planetary mechanisms may be larger or smaller than unity. The lower the value of this criterion, the higher, ceteris paribus, the reliability and durability of the planetary transmission.

6) The criterion of the relative speed of rotation of the locking clutches and disc brakes, which were switched off.

This criterion reflects the value of the maximum relative angular velocity of the master and slave disks of the "high-speed" clutch or disc brake, which were switched off. It allows you to assess the level of power loss, the amount of heating and deterioration of working drives in friction control elements, which were switched off. The value of this criterion is determined by the expression:

$$K_6 = \frac{|\omega_{LC(SB)max}|}{|\omega_{IV}|}, \quad (8)$$

where $\omega_{LC(SB)max}$ – the maximum relative angular velocity in the "high-speed" clutch or disc brake, which were switched off;

ω_{IV} – the angular velocity of the driving link of the planetary transmission.

7) The criterion a weighted average level of efficiency.

This criterion reflects the level of power loss in planetary transmission considering on the relative time of work on each gear. The value of this criterion is determined by the expression:

$$K_7 = \frac{1}{\sum t_i \eta_i}, \quad (9)$$

where $\sum t_i \eta_i$ – the sum of products of relative work time of the planetary gear on the i -th transmission and the calculated efficiency of the planetary transmission on this transmission.

8) Criterion average usage elementary planetary rows in the structure of planetary transmission.

This criterion reflects the average number of elementary planetary rows, which were considered in the analysis planetary transmission, involved in the power transmission. The value of this criterion is determined by the expression:

$$K_8 = \frac{n_G n_{EPM}}{\sum n_{LEPM}}, \quad (10)$$

where n_G – number of gears in the planetary transmission, with the exception of direct drive;

n_{EPM} – the number of elementary planetary mechanisms are considered when analyzing planetary transmission;

$\sum n_{LEPM}$ – the total number of loaded elementary planetary mechanisms on all gears in the planetary transmission, with the exception of direct drive.

The criterion K_8 assessed the degree of participation of elementary planetary mechanisms in the formation of all gears in the planetary transmission. Also, the criterion is estimated number of elementary planetary mechanisms operating at idle rotation without doing useful work in this valid and reducing the efficiency of the planetary transmission.

9) The criterion of the complexity of the device planetary transmission.

This criterion shows how much more in the planetary transmission is used control elements and the elementary planetary mechanisms than is necessary in the minimum configuration. The value of this criterion is determined by the expression:

$$K_9 = \frac{(n_{LC} + n_{SB} + n_{FW} + n_{EPM})}{(n_{LC} + n_{SB} + n_{EPM})_{\min}}, \quad (11)$$

where $(n_{LC} + n_{SB} + n_{FW} + n_{EPM})$ – the number of locking clutch – n_{LC} , stopping brake – n_{SB} , freewheel – n_{FW} and elementary planetary mechanisms – n_{EPM} , that were included in the planetary transmission which are considered; $(n_{LC} + n_{SB} + n_{EPM})_{\min}$ – number of locking clutches, brakes, and elementary planetary mechanisms, the minimum required for the synthesis of a simple planetary transmission, which is similar to the analyzed according to the number of degrees of freedom and number of gears.

10) The criterion of complexity of control system planetary transmission.

This criterion assesses the number of "on" and "off" the friction elements with an external control for consistent implementation all gears in the planetary transmission. If the scheme of planetary transmission provides controls with internal automaticity (freewheel), their switching "on" and "off" are not taken into account. This is done because they do not require external control and simplify the control system of the planetary transmission. The value of this criterion is determined by the expression:

$$K_{10} = \frac{(n_{LC,SB"ON"} + n_{LC,SB"OFF"})}{n_G}, \quad (12)$$

where $n_{LC,SB"ON"}$ – number of locking clutches – n_{LC} and brakes – n_{SB} , are switched “on” for each of the next gear in the planetary transmission; $n_{LC,SB"OFF"}$ – number of locking clutches – n_{LC} and brakes – n_{SB} , are switched “off” for each of the next gear in the planetary transmission;

n_G – number of gears in the planetary transmission.

When analyzing the quality criteria, some of which have been improved and new proposed allowing to more fully evaluate the degree of quality of scheme planetary transmission. Thus, the criterion of the loading of the stopping brake of planetary transmission proposed to define:

$$K_4 = \frac{M_{SB\max}}{M_{IN}}, \quad (13)$$

where $M_{SB\max}$ – maximum torque transmitted included stopping brake;

M_{IN} – torque supplied to input of the planetary transmission.

The criterion that determines the speed of rotation of the bearings placed in satellites proposed to define in two modes:

$$K_{5(1)} = \frac{|\omega_{sat\max(1)}|}{|\omega_{IN}|}, \quad (14)$$

where $\omega_{sat\max(1)}$ – the maximum angular velocity of the satellite in any planetary mechanism, that loaded torque;

ω_{IN} – the angular velocity of the driving link of the planetary transmission.

$$K_{5(2)} = \frac{|\omega_{sat\max(2)}|}{|\omega_{IN}|}, \quad (15)$$

where $\omega_{sat\max(2)}$ – the maximum angular velocity of the satellite in any planetary mechanism, that not loaded torque.

We have proposed a criterion of the loading of the brakes planetary transmission, which are involved in the braking of the machine:

$$K_{11} = \frac{M_{SB\max}}{M_{OUT}}, \quad (16)$$

where $M_{SB\max}$ – the value of the maximum torque at the brake which is loaded in the braking mode of the machine;

M_{OUT} – necessary for braking machines maximum braking torque is given to the output shaft of the planetary transmission.

We have also proposed a criterion for number axial layers (axial dimension). This criterion is the ratio between the total number of vertical lines and planes engaging gears on kinematic scheme at the level of the axes of the satellites and the number of elementary planetary mechanisms composing planetary transmission.

We have also proposed a criterion for number radial layers (radial dimension). This criterion is the ratio between the total number of horizontal lines in the saturated vertical section on kinematic scheme and the number of central links in one elementary planetary mechanism.

The proposed set of quantitative performance criteria allows to find the best kinematic scheme of planetary transmission in series under consideration. Here are various approaches to this issue.

If we assume that all of the proposed criteria are equally important, the overall integrated assessment of the scheme quality obtained by summing the values of these criteria.

If we assume that the quality criteria have not equal importance, we should use weighting coefficients. In the process of selecting weighting coefficients should respect the equality of their sum to one. If necessary, the weighting coefficients for certain criteria may be set to zero.

With this approach, the comparison of schemes for planetary gearboxes should be made for a sum of products of values of the criteria on their weighting coefficients.

An algorithm for constructing the kinematic scheme from the structural scheme.

After the creation of the structural scheme you need to make commutation of elements of the gearbox. This is due to the need to test the feasibility of the kinematic scheme.

The problem of modeling route (trace) is one of the most difficult tasks in the general problem of design automation. This is due to several factors, in particular the variety of ways of design and technological implementation of different routes, for each of which apply specific optimization criteria and constraints. From a mathematical point of view the trace - the most complicated task of choosing the optimal solution from a large number of options.

To perform commutation was developed the original algorithm. To find the shortest path between the elements was used wave algorithm Lee, which can be called one the most universal routing algorithms. Wave algorithm allows you to create a path between the two key points in any maze, where it can be formed. The only drawback of the algorithm is that it needs a large amount of memory.

The essence of the algorithm Lee can be summarized as follows:

Cells of start and finish are marked on the two-dimensional plaid card (matrix) consisting of a "passable" and "impassable" cells. The goal of the algorithm is the shortest route between the start cell and the finish cell, if possible. For this the wave propagates from the start in all directions. Each cell is traversed by the wave will be marked as "traversed". A wave cannot pass through the cells that where marked as "passed" or "impassable". The wave moves until it reaches the finish point, or until there are no more cells that wave is not passed. The path from start to finish is impossible to build if the wave has passed all of the available cells but is not reached the finish cell. The path is laid and stored in the array (passed by the coordinates of cells on the plane) after the wave has reached the finish cell.

For the tasks assigned originated the need for substantial improvements of the algorithm. For example, added the ability to find the shortest path between not only two points, and also between the polylines (point – element, polyline – link). The route will be laid between a pair of points belonging to the polylines, so that the path length is a minimum. To avoid unnecessary bends route have been introduced the priorities in the direction of movement. Priorities motion are chosen so that

the route has as less as possible bends. If you change the initial priority of movement you can get a variety of routes, but their length will be the same.

For the construction of the kinematic scheme searches a sequence of connections, that to allow all connections.

The most difficult for commutation are schemes, in which the components are connected to the driving and driven shafts, owned by one planetary gear that located between the other planetary gears, and schemes, in which communications is not carried out along the shortest path. (Fig. 1).

In the event that none of the connection cannot be made by the shortest route, we must allow to carry out the first connection is not on the shortest route. (Fig. 1).

For maximum efficiency of the algorithm all elements, which are simultaneously connected to the drive shaft (for example INPUT-11-32, fig. 1), are united in chains and each of them is taken as the input element (11 and 32).

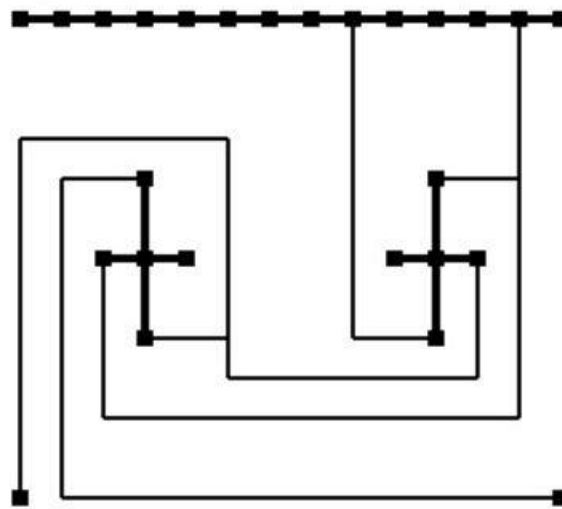


Fig. 1. Example of complicated scheme for commutation

Planetary mechanism is submitted on the kinematic scheme in the form of cross, as shown in Fig. 1, for $|k| = [1.5 \dots 8]$, or in the form of the letter "II" for $|k| = [1 \dots 1.5[$, as shown in Fig. 2, depending on the internal gear ratio.

Planetary rows presented in the form of a cross are elementary planetary rows or planetary rows with twin satellites (for $|k| > 4.5$). Planetary rows presented in the form of the letter "P" - planetary rows with two solar or two epicyclical gears. Planetary rows, presented in the form of the letter "II" at the time of commutation can roll over and turn around, to enhance the quality built kinematic schemes. Above and below the planetary rows left three lanes of communications, six between the gear sets, and three on each end.

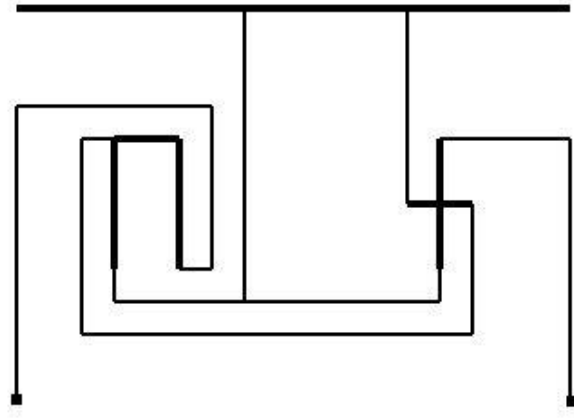


Fig. 2. Kinematic scheme with rows of different structures

To determine the order of connections you need to find a chain that contains the input element and to hold the first connection based on imposed limitations. Then all other communications are conducted in the order in which they appear in the code. If any link fails to hold, it is adjustable to a position above or to the position of number two (after the base connection with the restriction). This continues for as long as it will be carried out all connections, or two connections begin to change places with each other. In the second case, the first connection performed with other restriction and the algorithm is repeated. This continues until it is able to hold all connections, or options constraints is not end. If the connection fails, it is believed that the kinematic scheme cannot be implemented.

Description of the program and its interface.

The program's interface is shown in Fig. 3.

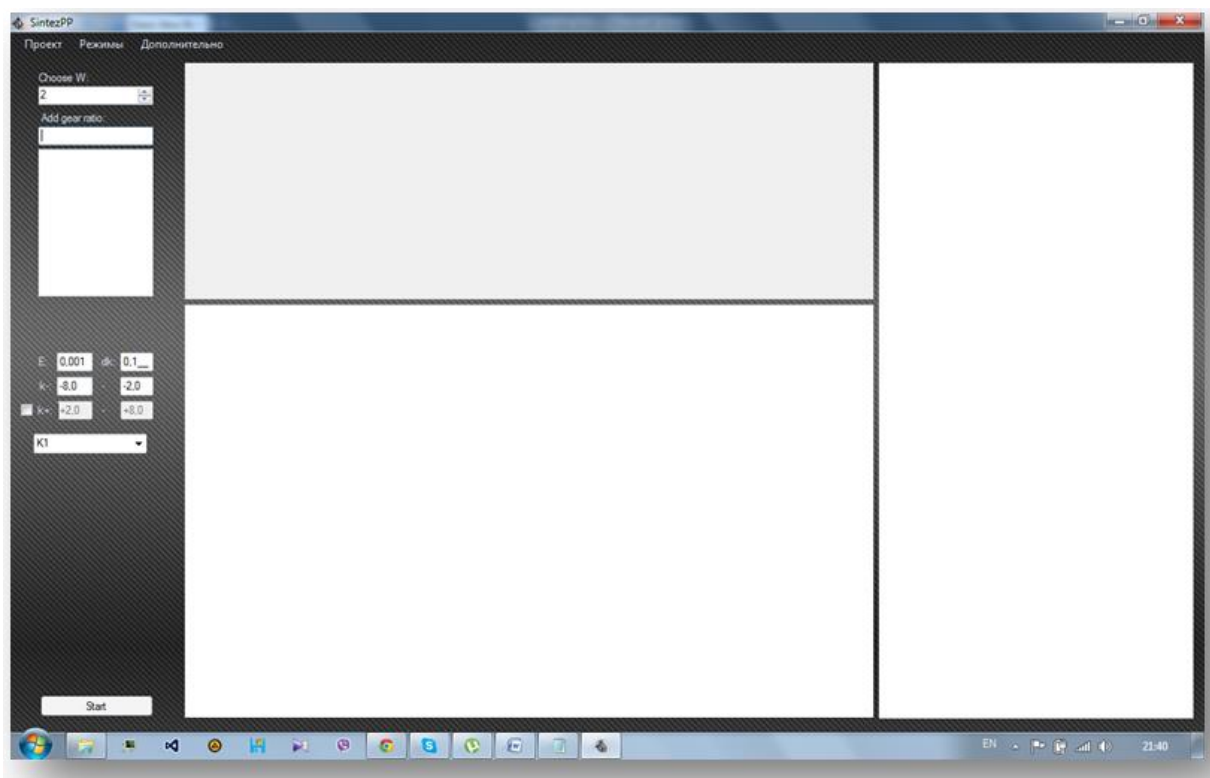


Fig. 3. Interface of program

After starting the program opens the main form. On the form, you need to choose the number of degrees of freedom of the control panel. Then you need to add the gear ratios.

If the user desires to independently specify which elements are connected to the input and output shafts, it should select "Generate with In and Out" in the program menu. If necessary, you can change the search options internal gear ratios. Then press the "Next" button and wait until you see the results of calculations (Fig. 4).

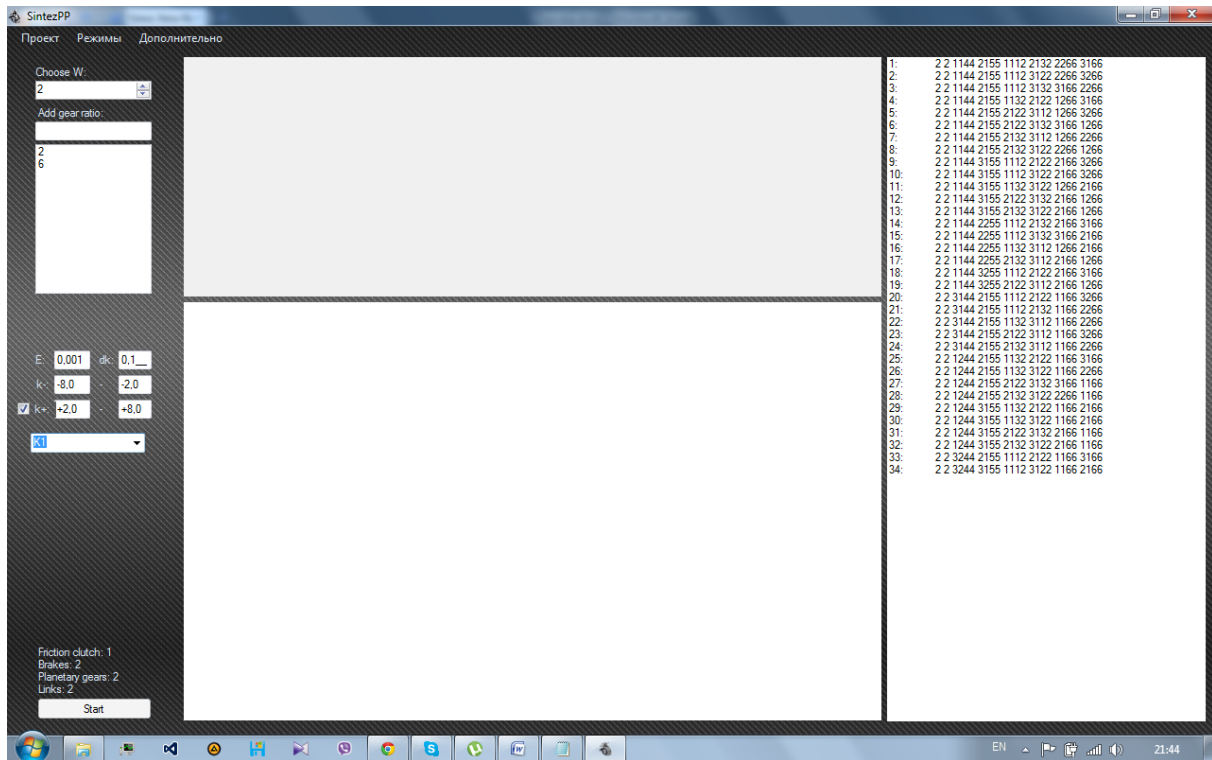


Fig. 4. Codes generated schemes

To view a particular scheme you must click on her arm. After this the screen will show it's block diagram and in the field underneath – the results of the kinematic analysis (Fig. 5). To select the best scheme, the program provides the ability to sort schemes by criteria of quality.

The results of the program can be saved to a file. To do this, you must perform a "Project -> Save" and choose a directory to save. To open the saved calculations, you must perform a "Project -> Load". It is also possible to save the results of calculations in Excel-file, it is necessary to execute "Project -> Save as XLS».

You also have the possibility of calculating the particular scheme. To use this module of the program, the operator must make himself a block diagram.

To start the mode of calculation of a particular scheme, you must select the manual mode in the menu and enter gear ratios. In the graphics area will appear icons of planetary ranks without connections. Then user must select in the graphical field the pairs of elements that it wants to attach with each other (Fig. 6). When the user has set all the necessary connections, except for the brakes, the program offers to determine the elements that are connected to the brakes automatically. This is possible because in a planetary transmission with two degrees of freedom the position of the brake is uniquely determined. Pairs of connections are written to the vector, which has the following structure:

X – the amount of basic planetary mechanisms;

X – the amount of brakes;

XX44 – element, which connected to the driving shaft;

XX55 – element, which connected to the driven shaft;

XXXX – connections;

...

XX66 - element, which connected to the brakes;

....

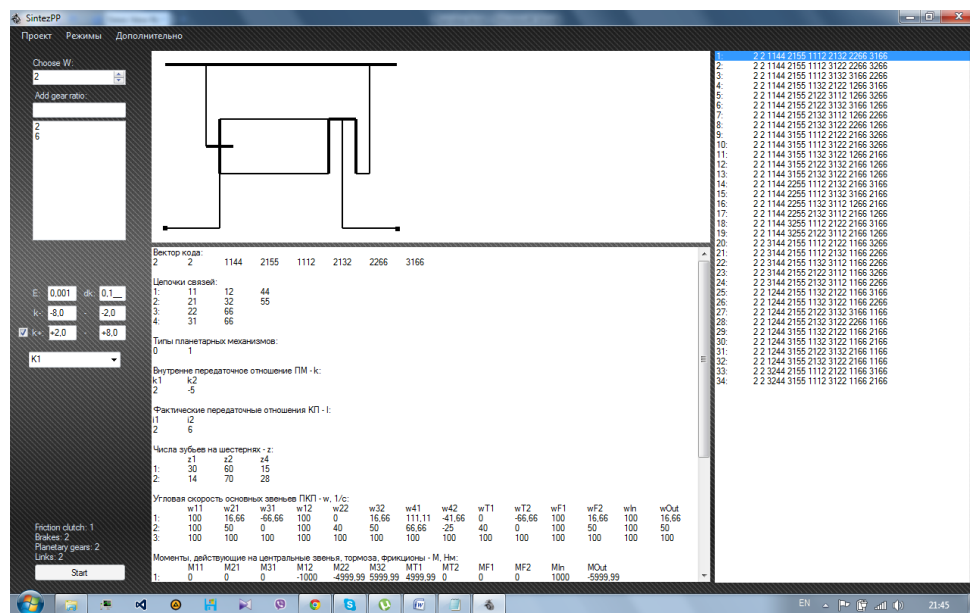


Fig. 5. View of calculation results for a particular scheme

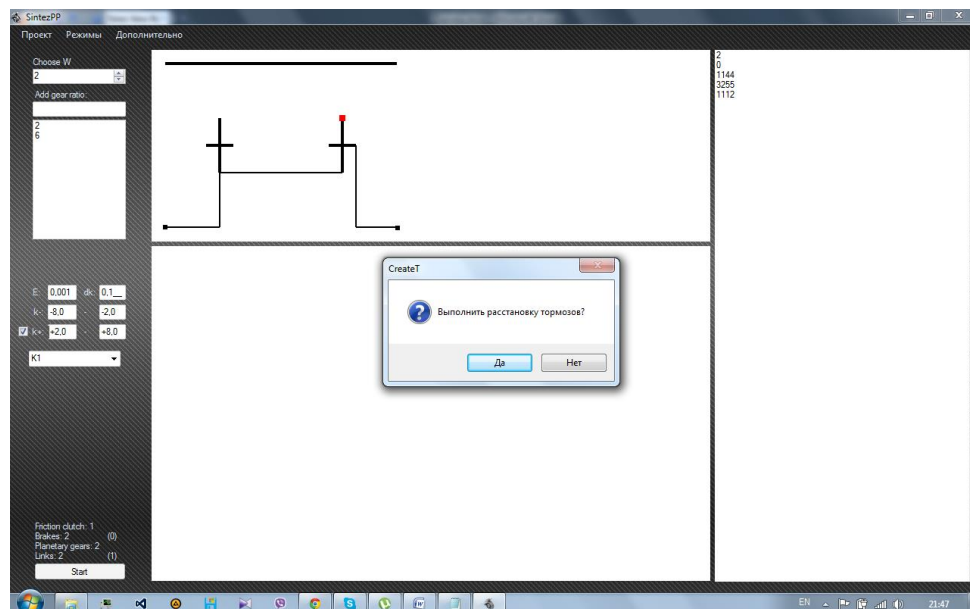


Fig. 6. Example of calculation of particular scheme

If at some stage, the calculations turned out that to implement the scheme cannot be given, the program displays an appropriate message.

Conclusion. As a result of this work proposes an algorithm and software that make possible according specify gear ratios [12] in automatic mode to obtain a kinematic scheme of a planetary transmission with two degrees of freedom, with an arbitrary structure of planetary mechanisms, which satisfies all design constraints, and has the best performance on selected quantitative quality criteria.

As a further research is planned to expand the capabilities of the algorithm and software on planetary transmission with three degrees of freedom.

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